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Modular Synthesis of Plane Lever Six-Link Mechanism of High Class

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Abstract: Advantages of the modular synthesis method are considered based on an example of the problem of synthesis of six-link hinge-lever mechanisms. In the case where four-link mechanism cannot reproduce a predetermined motion of the output link with satisfactory accuracy, then the synthesis of six-link mechanisms is considered. Herein, the solution of the assigned task of synthesis of six-link transfer mechanisms is reduced to approximated problem of finding of umbilical points and has an analytical solution for three of the five parameters, so the search for the minimum of the objective function is performed by two variables only. This tangibly facilitates the computational algorithm.

Key words: Six-link · Hinge-lever mechanisms · Four-link mechanism · Algorithm

INTRODUCTION

As you know, it is more appropriate applying block and modular synthesis methods in terms of the requirements of the automation of conceptual design stage of mechanisms. In this case, the procedure of mechanisms synthesis with different structural scheme and different classes may be reduced to the uniform procedures of structural modules synthesis - the simplest structural units of synthesis [1].

The transfer function is desined during transfer mechanisms synthesis $\psi = \psi(\varphi)$, $\varphi_1 \le \varphi \le \varphi_2$, where by φ is designated generalized coordinate of a mechanism and by ψ - turning angle of balancing lever (Fig. 1). In the tasks of practical synthesis which are solved using numerical methods, the transfer function is defined by N_S and successive values ψ_k^* in tabular form. For automated synthesis of mechanisms of the second class – it isdeveloped a program of synthesis “Opt_syn” with a number of links 4, 6 and 8 (program segments are given in the Annex) wherein the numerical minimization of objective function is made:

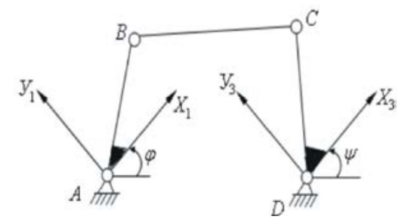

Fig. 1: Hinge four-link chain as a transfer mechanism

$$f(P) = \frac{1}{N_S}\sum_{k=1}^{N_S}\left[\eta_k \Delta_k(P)\right]^{2m} \Rightarrow \min_P f \quad (1)$$

or

$$f(P) = \max_{k=1,\ldots,N_S} \left|\eta_k \Delta_k(P)\right| \Rightarrow \min_P f \quad (2)$$

here Δ_k - is a deviation function

$$\Delta_k(P) = \psi_k(P) - \psi_k^*,\ k = 1,\&, N_S \quad (3)$$

Defining the parameters of the synthesis of P is carried out by minimizing the presented objective functions by well-known numerical methods:

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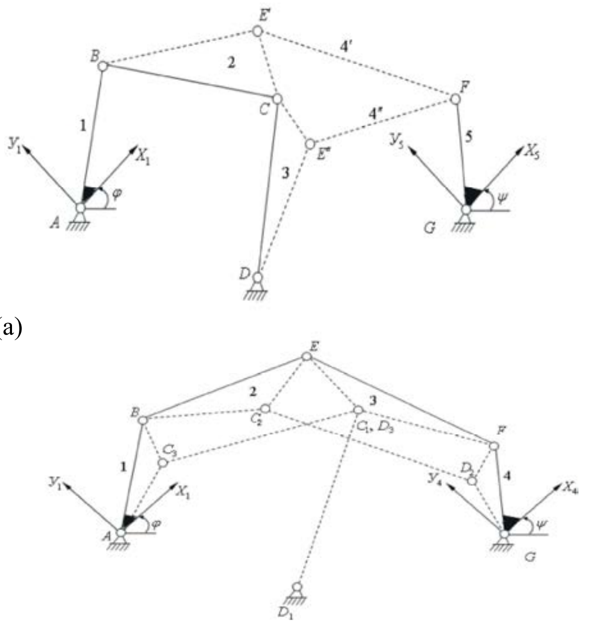
- Nuldera-Midda method of deformable polyhedron for the function of many variables (simplex algorithm);
- Quasi-Newton method of zero-order to determine the absolute minimum with the assessment of values of the first and second derivatives of the objective function, which depends on several arguments;
- Combined method of Gauss-Newton and modified Newton's method for the function of several variables (specialized high-performance algorithm for finding the minimum of the function, which is the sum of squares m of nonlinear functions of n variables, where $m \geq n$).

The program is based on the repeated analysis of the provisions of the mechanism - a problem of provisions analysis is solved by a computer as many times as references are made to the procedure of calculating the value of the objective function. Note, that in this case, to perform a single step to the minimum descent - algorithm can perform multiple references to the procedure of calculating the value of the objective function. In this regard, there are certain computational complexities during the synthesis of mechanisms of the third, fourth and high classes, because the task to analyze the provisions for such mechanisms is conjugated with a complex algorithm of identification of mechanism assemblies[2].

Model Structure: Advantages of modular synthesis method can be shown on example of the synthesis problem of six-link hinge-lever mechanisms. In the case where four-link mechanism cannot reproduce a predetermined motion of the output element with a satisfactory accuracy, then is considered a synthesis of six-link mechanisms. In this case, the number of possible structural solutions of this problem is equal to 5: two second-class mechanisms and three high-class mechanisms. In design engineering practice, the mechanisms of the second class are still mainly used, rarely mechanisms of the third class and mechanisms of the fourth class are not used at all. Thus, only 40% of the available arsenal of structural solutions finds a practical use. If the solutions obtained on the basis of the synthesis of six-link mechanisms of the second class do not meet the designer, then, usually, is considered a synthesis of eight-link mechanisms of the second class. Thus, the functional capabilities of high-class mechanisms are not used and the main reason for that – is the computational complexity specified above.



(a)

(b)

Fig. 2: Structural synthesis of six-link transfer mechanisms of the second (a), third and fourth classes (b).

Figure 2 shows the structural synthesis of six-link transfer mechanisms of the second class. Kinematic chain *BCD* is connected to the input link 1 and stand 0, then by connecting a binary link 4 to the output link 5 - six-link mechanisms are formed. Herein, two alternate versions of the binary link connection are possible: in the first case, a binary link 4 connects the output link 5 with a rod 2 and in the second case - with balancing lever 3. In either case, the standard procedure of synthesis of the binary link 4 is carried out. When varying the parameters of *BCD* dyad, it is necessary to enter an additional constraint - ensuring a predetermined limit of the pressure angle. The same additional requirement is placed on the pressure corner for *EFG* dyad. Therefore, in the case of synthesis of the second class mechanisms – the provision of favorable force transmission from the input link to the output link and work of dyadic groups in the selected assembly is carried out without much difficulty [3].

However, this is not true in the case of synthesis of the third and fourth class mechanisms. The main reason - a lack of geometric criteria for evaluating the quality of motion transmission in multi-loop high class mechanisms. Fig. 2 b shows the structural synthesis of the third and fourth class mechanisms. Input link 1 and output link 4 are connected by *BEF* dyad, then again a standard operation of binary link 5 synthesis is performed in one of three alternate versions: 1) binary link C_1D_1 connects link 3 with a post – the three-point third class mechanism is formed; 2) binary link C_2D_2 connects link 2 with the output link 4 – the fourth class mechanism is formed; 3) binary link C_3D_3 connects link 3 with the input link 1 – the two-point third class mechanism is formed [2, 4, 5, 7].

Thus, the following advantages are achieved in modular synthesis:

- The synthesis problem is solved by the use of a unified algorithm applied both to the mechanisms of the second and mechanisms of the high classes. Thus, in the synthesis of six-link transfer mechanisms, the problem is reduced to the synthesis of the binary link.
- Synthesis operation is reduced to the problem of structural module synthesis (the smallest structural unit of synthesis), for which there is an analytical solution regarding the variables. Thus, in the synthesis of six-link transfer mechanisms, the approximation problem of finding the umbilical points has an analytical solution for three of the five parameters, so the search for the minimum of the objective function is carried out per two variables only. This significantly facilitates the computational algorithm [4].
- There is no need to make multiple analysis of the mechanism, i.e. to solve a complicated problem of analyzing provisions. The use of indirect criterion in synthesis – is a structural error related to the synthesized structural module and an analysis of the provisions with the definition of the true error at the output of the mechanism is performed at the end of the synthesis procedure. Absence of mechanism discontinuity and remoteness from the singular provisions is ensured through the use of additional criteria constraints [5].

Experimental Results: Meeting of the latter requirement is made possible through the use of introduced analog of angle of motion transmission for the high class mechanisms.

Example 1: The fourth-class mechanism with a motion dwell of output link with the duration of motion dwell of 180° is obtained as a result of synthesis. Dimensions of obtained mechanism are given below:

Parameter symbols	Values
L1 (AB)	0,188
AG	1
L2 (BC)	0,5284
L2_1 (BD)	1,6022
beta2	-267,2754
L3 (CF)	0,2487
L4 (DE)	1,5504
L5 (GF)	0,6345
L5_1 (EG)	1,0301
beta5	171,4746

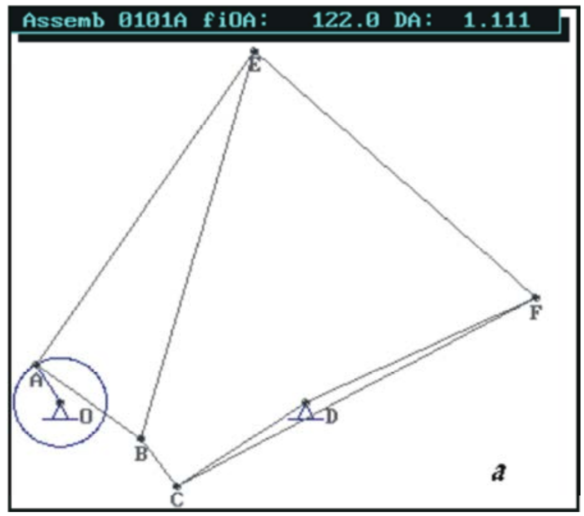


Fig. 3: Graphic chart of mechanisms from the computer screen for the 1st mechanism assembly

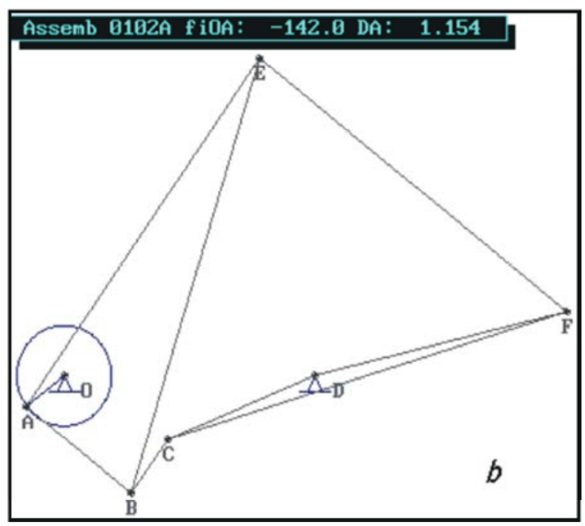


Fig. 4: Graphic chart of mechanisms from the computer screen for the 2nd mechanism assembly.

Figure 3 and 4 demonstrate graphic charts of mechanisms from the computer screen for two mechanism assemblies.

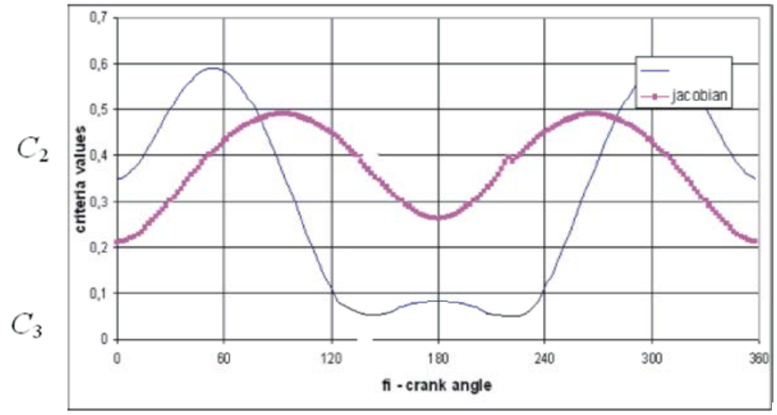


Fig. 5: Change in the indexes of motion transmission c_2 and c_3 for the 1st assembly

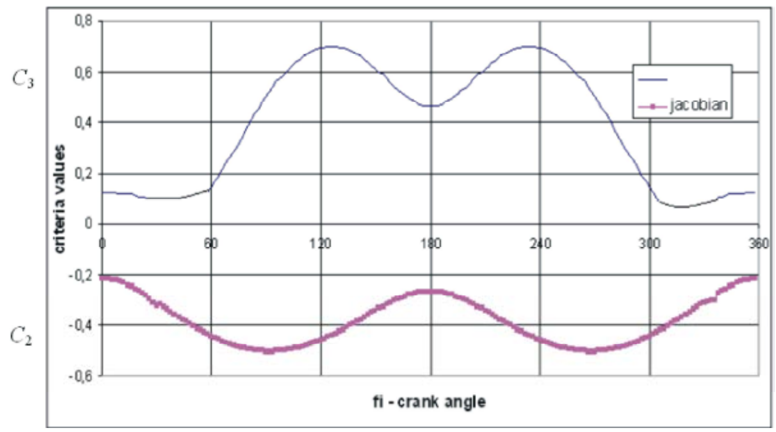


Fig. 6: Change in the indexes of motion transmission c_2 and c_3 for the 2nd assembly

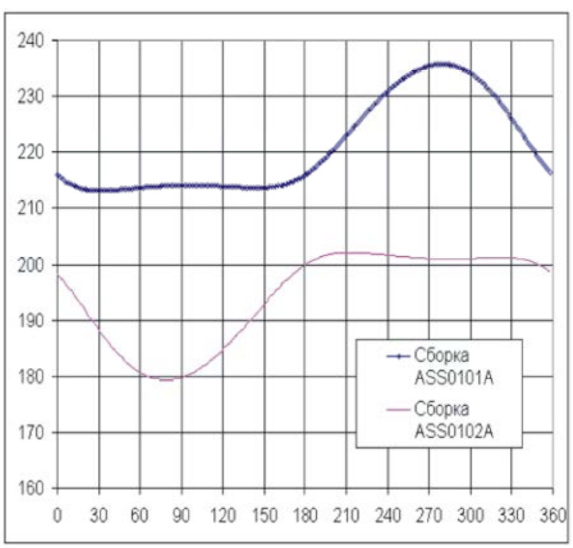


Fig. 7: Functions of positions with 2 assemblies of transfers

Figure 5 and 6 demonstrate changes in indexes of motion transmission c_2 and c_3 for two assemblies of mechanisms, respectively. You can see decrease of c_2 when c_3 decreasing. But increase of index c_3 does not mean the increase of c_2 [6].

Functions of positions with 2 assemblies of transfers are shown on Figure 7.

Example 2: The fourth-class mechanism with a motion dwell of output link with the duration of motion dwell of 180° is obtained. Dimensions of obtained mechanism are given below. Figure 8, 9, 10 demonstrate graphic chart of mechanisms from the computer screen, change in the indexes of motion transmission c_2 and c_3 and function of position for mechanism, respectively:

Parameter symbols	Values
L1 (AB)	0,1295
AG	1
L2 (BC)	0,4212
L2_1 (BD)	0,7979
teta2	-264,9023
L3 (CF)	0,2173
L4 (DE)	0,4339
L5 (GF)	0,8528
L5_1 (EG)	0,0964
teta5	301,8164

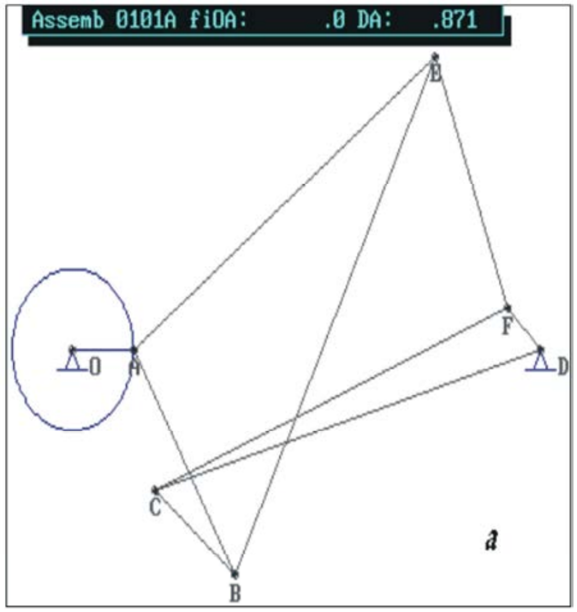


Fig. 8: Graphic chart of mechanism from the computer screen for mechanism of example 2

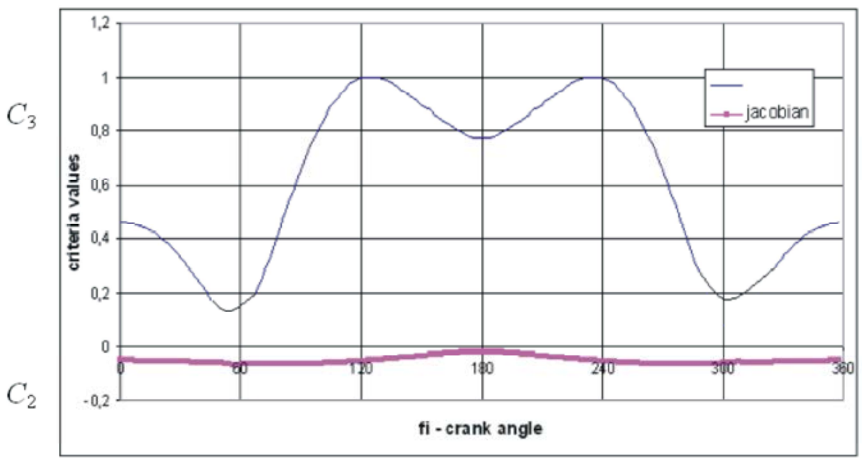


Fig. 9: Change in the indexes of motion transmission c_2 and c_3 in example 2

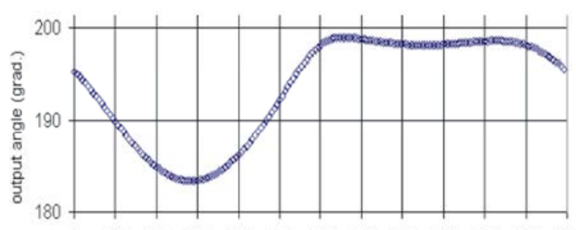


Fig. 10: Function of position for mechanism of example 2

Example 3: The fourth-class mechanism with a motion dwell of output link with the duration of motion dwell of 150° is obtained. Dimensions of obtained mechanism are given below. Figure 11, 12, 13 demonstrate graphic chart of mechanisms from the computer screen, change in the indexes of motion transmission c_2 and c_3 and function of position for mechanism, respectively [7]:

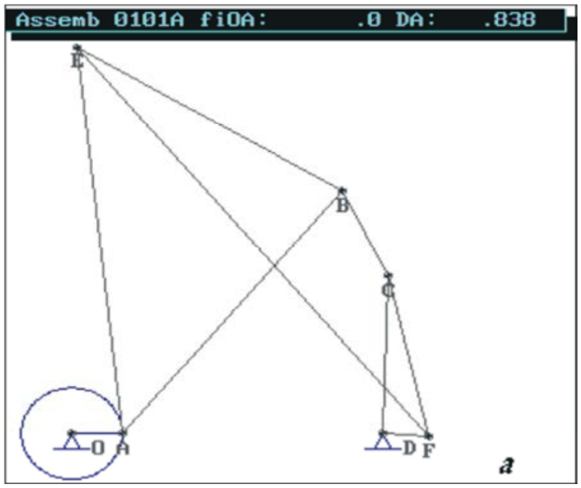


Fig. 11: Graphic chart of mechanism from the computer screen for mechanism of example 3

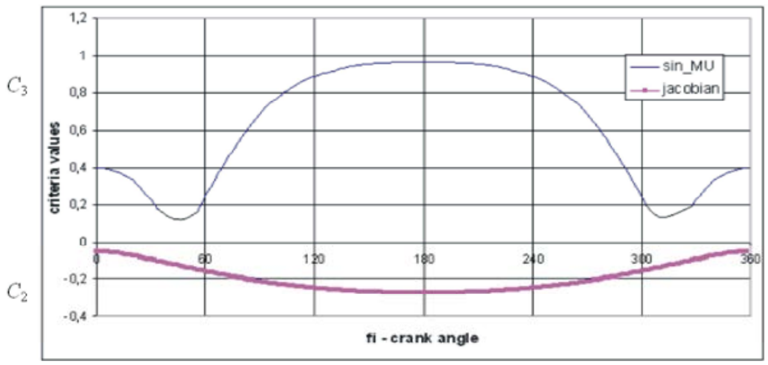


Fig. 12: Change in the indexes of motion transmission c_2 and c_3 in example 3

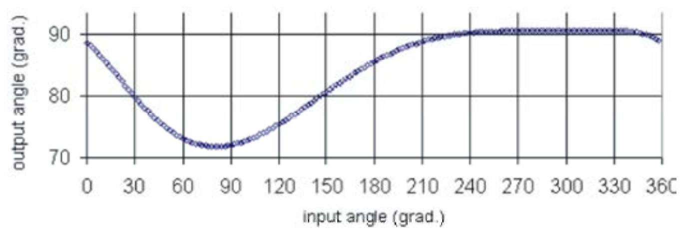


Fig. 13: Function of position for mechanism of example 3

Parameter symbols	Values
L1 (AB)	0,1625
AG	1
L2 (BC)	1,1194
L2_1 (BD)	1,3937
teta2	45,2637
L3 (CF)	0,3362
L4 (DE)	1,7981
L5 (GF)	0,5651
L5_1 (EG)	0,1451
teta5	266,5723

CONCLUSION

Thus, the following results will be obtained in modular synthesis:

- A unified algorithm is developed which allows the solution of synthesis problems of mechanisms of both the second and high classes.
- An approximating problem of finding the umbilical points has an analytical solution for three of the five parameters in the synthesis of six-link transfer mechanisms.
- An introduction of additional criteria restrictions significantly facilitates the solution of the assigned complex problem of synthesis, which is achieved by the introduction of analog of motion transmission angle for high class mechanisms. Dimensions of obtained mechanism are given.

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